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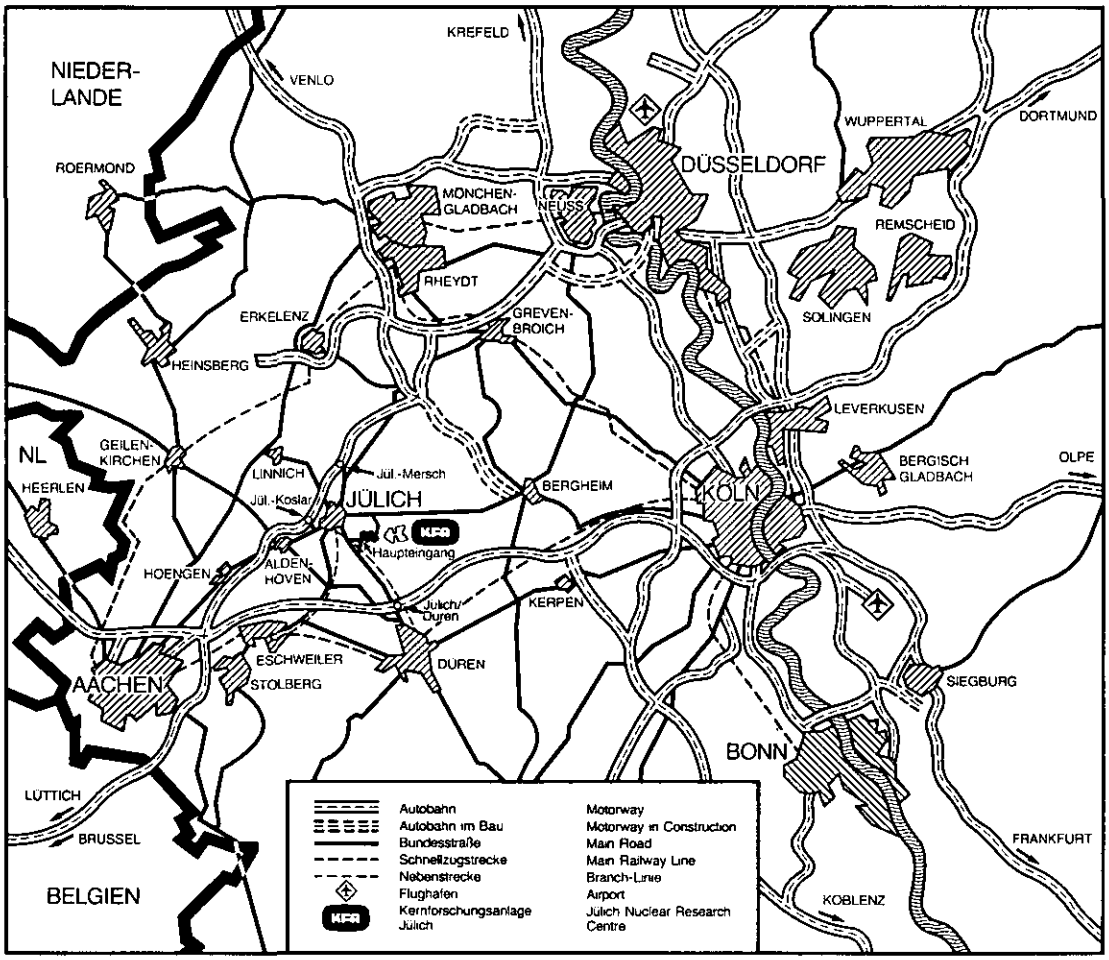

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A Program Module Simulating a Gas-heated Steam Generator with Steam Condensation in the Primary Flow Channel

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Abstract

A program module, written as a FORTRAN subroutine, is described which simulates the behaviour of a countercurrent steam generator heated on the primary side by a mixture of a noncondensable gas and water steam. The module simulates possible steam condensation in the primary channel. The routine which has been optimized with regard to computing time requirements can be used, for instance, as a module in computer programs simulating water ingress accidents in high temperature reactors.

Ein Programm-Modul zur Simulation eines gasbeheizten Dampferzeugers mit Dampfkondensation im primären Strömungskanal

von

G. Meister

Zusammenfassung

Es wird ein Programm-Modul beschrieben, welches das Verhalten eines Gegenstromdampferzeugers beschreibt, welcher auf der Primärseite durch ein Gemisch aus einem nicht kondensierbaren Gas und Wasserdampf beheizt wird. Das Modul simuliert Dampfkondensation im Primärkanal, falls die Voraussetzungen dafür gegeben sind. Es ist als FORTRAN-Unterprogramm geschrieben und im Bezug auf den Rechenzeitbedarf optimiert. Es kann z.B. als Baustein eines Rechenprogramms benutzt werden, welches Wassereinbruchstörfälle bei Hochtemperaturreaktoren simuliert.

Acknowledgement

The author is indebted to Mr. H. Kawamura from JAERI, Japan, for numerous clarifying discussions on modelling of condensation phenomena during his stay as a guest scientist at the KFA. I am also grateful to Mr. Saphier from the SOREQ nuclear research center, Israel, for suggesting to apply the variable boundary technique to steam generator simulation.
The manuscript has carefully been written by Mrs. Tietz.

Contents

Introduction

Heat removal in power plants with high temperature reactors is achieved via steam generators which are heated on the primary side by helium flow. Inlet temperatures under normal operating conditions are in the range from about 850°C to 950°C.

Large power plants are equipped with several heat removal loops which all have the same performance and are operated in a parallel mode. Six loops, for instance, are used in the THTR plant.

The design of the large heat exchangers used for plant operation at full power is of the helix bundle type. Characteristic for this design are heat exchanging tubes subject to cross flow of the heating gas.

The primary gas pressures in high temperature reactors are in the range of about 40 bar, while the pressure on the secondary side of the heat exchangers is kept at fairly high values. Secondary steam pressures in the range from 100 to 200 bar are typical. In the case of a tube rupture in one steam generator, therefore, water or steam will ingress into the primary circuit. If the ingress is not interrupted by some countermeasure large amounts, up to several tons of water may enter the primary system. In this case a pressure increase may result which endangers the integrity of the pressure vessel.

Regarding this problem the question arised whether condensation of circulating steam could reduce the rate of pressure increase if the steam partial pressure becomes sufficiently high.

If the secondary feed water supply will continue at least in the intact steam generators, condensation may be expected to appear first in the economizer part. Since the surface temperature in this part is determined by the relative low feed water inlet temperature, the economizer tubes will form the coldest part of the surfaces in the primary circuit under these accident conditions.

The study presented here investigates possible condensation effects in typical HTR steam generators and describes a computer code which can be used to quantify the influence of this effect on the plant response.

1. Applicability of the code and restrictions

The program module which is described in this report simulates the behaviour of a steam generator with countercurrent flow in the primary and secondary channel. The primary fluid is admitted to be a mixture of water steam and noncondensable gas. Possible condensation of steam on the surfaces of the primary flow channel is included into the model underlying this code.

The module has been developed to combine low computing time requirements with reasonable accuracy when applied to the simulation of water ingress accidents in HTR. It has successfully been used as a subroutine in the code COROX /1/ which simulates the dynamic behaviour of the primary circuit of a high temperature reactor under accident conditions. Since this code, when used for the analysis of long term transients, may call the steam generator module several hundred times per run, it is evident that computing economy becomes an important aspect.

The module is based on a quasi steady state approach. This means that any energy storage in the structure and in the fluid is neglected. The module, therefore, should not be used to simulate fast transients where such storage effects contribute essentially to the systems response. Other simplifications in the treatment of the condensation mechanisms in the primary flow channel have been suggested by the results obtained by H. Kawamura and G. Meister /2/ with a more detailed condensation model.

The simplifications introduced may, of cause, result in a certain loss of accuracy but comparative calculations with detailed steam generator codes have shown that deviations are in general small enough to be acceptable.

2. Partitioning of the heat exchanging area

The evaluation technique used in this module is a variable boundary method. The number of intervals of the heat exchanging area over which the energy balances are integrated in one step is reduced to the actual minimum required

by the flow conditions in the primary and secondary channel. The method is characterized by interval boundaries which are not fixed but depending on inlet conditions.

The total heat exchanging area of the steam generator is subbdivided into three parts as shown in Fig. 1. They correspond to the subcooled region, the evaporation region and the steam superheating region on the secondary side. In the subcooled region single phase water flow is assumed. Any steam generation by subcooled boiling, therefore, is neglected. Single phase flow is also assumed in the steam superheating region which means, that an entrainment of water droplets into the steam flow is not taken into consideration.

The location of the boundary of the subcooled water region is determined by that point, where the water enthalpy attains the saturation enthalpy of water corresponding to the pressure at the secondary side. The boundary between the evaporation region and the steam region is given by the saturation enthalpy of steam. In the evaporation region, therefore, the fractional steam mass flow (steam quality) varies between zero and one.

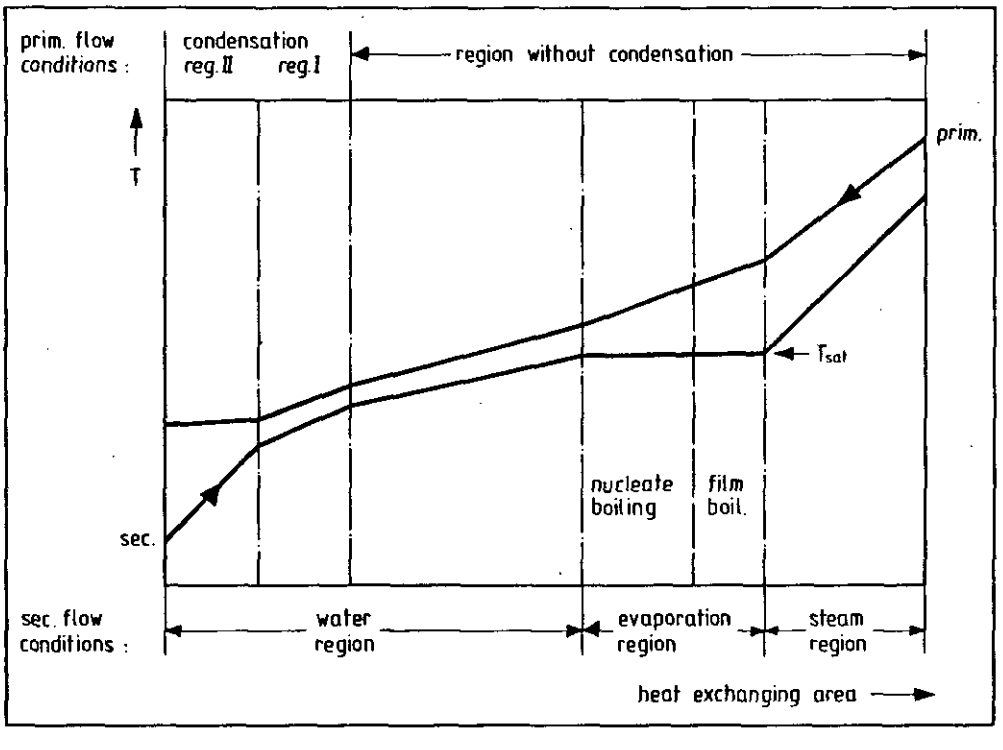

Fig. 1: Scheme of variable boundary steam generator model.

Thermal equilibrium is assumed throughout the evaporation region. The fluid temperature, therefore, is assumed to be constant and equal to the saturation temperature.

The evaporation region is subdivided into two subregions with different heat transfer conditions on the secondary side. In the first subregion (adjacent to the water region) nucleate boiling heat transfer is assumed and in the second subregion film boiling heat transfer. The boundary between these two subregions is given by a value of the steam quality at which departure from nucleate boiling (DNB) is expected to occur. The corresponding fluid enthalpy, therefore, is

$$h_{2DNB} = h_{satw} + L \cdot x_{DNB} , \qquad (1)$$

where h_{satw} is the saturation enthalpy and L the latent heat of water.

The DNB-quality is evaluated from empirical correlations. The film boiling subregion is, of course, missing if the DNB-quality is equal to one.

The subcooled region, in turn, is subdivided into three subregions with different conditions on the primary side. The first region is characterized by simultanous wall and bulk condensation (condensation region II). The bulk temperature in this region is equal to the saturation temperature corresponding to the local steam partial pressure. The next subregion is characterized by a bulk temperature above the saturation temperature but with the wall temperature equal to or below it. This region exists if heat flow is directed from the primary to the secondary side and, therefore, the wall temperature is lower than the bulk temperature. Finally, the third subregion is characterized by the absence of condensation because bulk and wall temperature are both higher than the saturation temperature. The temperature profiles of the primary flow which give rise to the existence of the different condensation regions are schematically shown in Fig. 2. We note that any condensation on the primary side in the range of evaporation and superheating on the secondary side is out of consideration. It can easily be shown that this tacit presupposition is always satisfied under the conditions in question.

The location of the boundary between the condensation region I and the region without condensation is given by the point where the primary wall temperature attains the saturation temperature related to the inlet steam partial pressure. The boundary between condensation regions I and II is determined by the point where the primary bulk fluid temperature attains the saturation temperature according to the local steam partial pressure.

The locations of the different boundaries depend on flow and inlet conditions. Therefore, they may vary during a transient. One or more of these boundaries may be missing under certain inlet conditions. The maximum number of regions defined by these boundaries is six but the number of regions actually present may be smaller.

The variable boundary concept requires as a necessary presupposition to be applicable constant values for certain quantities within every region. So far as these quantities vary in reality, they have to be replaced by some suitable average which, for instance, can be evaluated for the initial condition from a more detailed steam generator program.

For the secondary circuit, suitable values of the effective specific heat capacity at constant pressure and the heat transfer coefficient must be attributed to every of the four regions of the secondary flow.

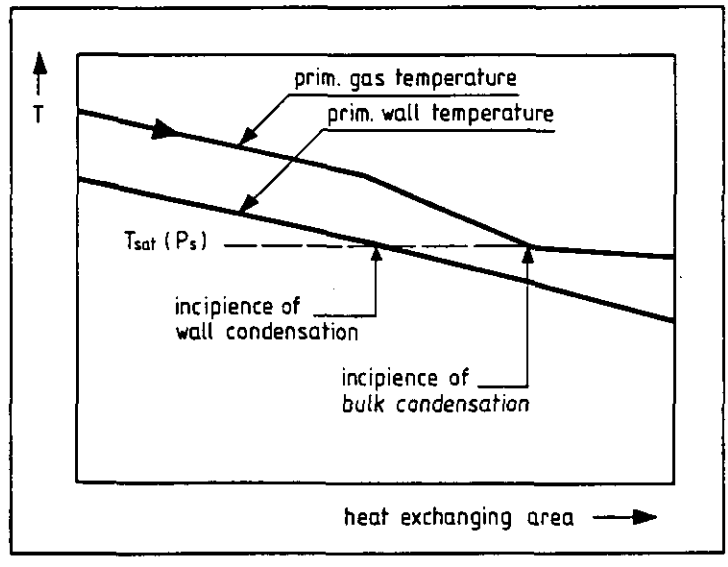

Fig. 2: Schematic representation of condensation regions in the primary flow channel.

3. Governing equations

3.1 Mass balances

From the steady state continuity equations follows that the total mass flow in the primary and in the secondary flow channel are constant throughout the steam generator. The primary mass flow is assumed to consist of a water component and a component of noncondensable gas

$$m_1 = m_{H_2O} + m_{nc} \tag{2}$$

The latter may, for instance, be helium containing impurities such as oxygen, nitrogen and other gases resulting, for instance, from graphite corrosion. Both components of the mass flow are constant because no mass exchange is possible. The water mass flow, in turn, is assumed to consist of a steam fraction and a condensed fraction. The latter is assumed to consist of small water droplets carried with the main gas stream (fog flow) and, if wall condensation occurs, of a certain fraction of condensate flow in a water film on the wall surface. The total water flow, therefore, may be written

$$m_{H_2O} = m_s + m_w + m_f \tag{3}$$

The three contributions may vary during their passage through the primary channel. Mass transfer between the steam fraction and the water fraction may arise due to condensation or evaporation. Mass transfer between film and bulk flow may be due to entrainment of the wall condensate into the bulk in form of water droplets.

The thickness of the condensate film on the surface of the heat exchanging wall is assumed to be in equilibrium, i.e. constant in time under otherwise steady state conditions. This means that no net water transfer exists between the bulk and the film: any water formation on the wall by condensation is compensated by entrainment to the bulk. Since, in addition, the film mass flow remains small compared with a total mass flow, the splitting of the water flow into a droplet flow and a film flow which has been included into the model of ref. /2/ has been abandoned in this module.

The gaseous components of the primary flow are approximated as ideal gases satisfying Dalton's law:

$$p_g = p_{nc} + p_s \tag{4}$$

where p_g is the total pressure. Since pressure loses are neglected, p_g is a constant quantity throughout the primary channel.

The gas densities are given by

$$\rho_{nc} = M_{nc}\, p_{nc}/(R\, T_1) \quad ; \quad \rho_s = M_s\, p_s/(R\, T_1) \tag{5}$$

where M_{nc} is an effective molecular weight of the mixture of noncondensable gases. M_s is the effective molecular weight of steam

$$M_s = 18/Z_R \tag{6}$$

where Z_R is a correction factor which takes the real properties of steam into account. Z_R is a function of steam pressure and temperature. It is essentially different from 1 for steam pressures larger than about 1 bar and for moderate steam temperatures. The effective molecular weight of steam may be supplied by the calling program according to the steam partial presssure at the primary inlet and for some average steam temperature. The variation of M_s due to temperature and pressure changes along the primary channel, however, is not taken into account.

If all gas components have the same flow velocity and temperature it follows for the steam mass flow

$$m_s = (m_1 - m_w)\, M_s\, p_s/(M_{nc}\, p_{nc} + M_s\, p_s) \tag{7}$$

and for the flow of the noncondensable gas

$$m_{nc} = (m_1 - m_w)\, M_{nc}\, p_{nc}/(M_{nc}\, p_{nc} + M_s\, p_s) \tag{8}$$

Bearing in mind that m_{nc} is a constant quantity, m_s may be written

$$m_s = m_{nc} (M_s/M_{nc}) p_s/(p_g - p_s) \quad (9)$$

This equation shows the dependence of the steam mass flow on the local steam partial pressure.

3.2 Energy balances

From a steady state energy balance for the primary and secondary flow follows

$$\Delta H_1 + \Delta H_2 = 0 \quad (10)$$

where ΔH_1 and ΔH_2 are the changes of the primary and secondary enthalpy flow between inlet and outlet. Since kinetic energy terms are neglected H_2 may be written

$$H_2 = m_2 h_2 \quad (11)$$

where h_2 is the enthalpy of the secondary fluid. The correspondig expression for the primary flow is

$$H_1 = m_{nc} h_{nc} + m_s h_s + m_w h_{1w} + m_f h_{1wf} \quad (12)$$

where h_s is the steam enthalpy, h_{1w} the water enthalpy in the bulk and h_{1wf} is the water enthalpy in the film.

In the bulk flow thermal equilibrium is assumed. If no condensation takes place, all components of the bulk flow, therefore, have the same temperature. The change of primary enthalpy flow may then be written

$$\Delta H_1 = (mc_p)_1 \Delta T_1 \quad (13)$$

with

$$(mc_p)_1 = m_{nc} c_{pnc1} + m_s c_{ps1} + m_w c_{pw1} \quad , \quad (14)$$

and the energy balance (10)

$$m_2 \, (h_{2,i} - h_{2,i-1}) = - (mc_p)_1 \, (T_{1,i} - T_{1,i-1}) \qquad (15)$$

where the indices i and i-1 denote two points of the heat exchanging area which do not include condensation on the primary side.

In the case of volume condensation, the steam partial pressure and thus the steam mass flow varies in the primary channel. Due to the presupposition of thermal equilibrium, all gas components in the bulk will also have the same temperature which, in this case, is equal to the saturation temperature corresponding to the local steam partial pressure.

If the contribution of the film enthalpy flow is neglected the following expression for the change of the primary enthalpy flow is obtained

$$\Delta H_1 = (mc_p)_1 \, \Delta T_1 + L \, \Delta m_s \quad \text{with} \quad T_1 = T_{sat}(p_s) \qquad (16)$$

where

$$L = (h_s - h_{1w})_{T=T_{sat}} \qquad (17)$$

is the latent heat of condensation corresponding to the local steam partial pressure.

3.3 Heat exchange between primary and secondary flow

The energy equation (10) may be broken up into the following two equations

$$dH_1/dz = - S_1 \, q_1 \quad ; \quad dH_2/dz = S_2 \, q_2 \qquad (18a, b)$$

with

$$S_1 \, q_1 = S_2 \, q_2 \qquad (19)$$

q_1 and q_2 are local heat flux densities and S_1, S_2 the heat exchanging perimeters of the primary and secondary flow channel respectively. In the case of round tubes with the secondary fluid flowing inside, the following relation holds

$$S_1/S_2 = r_1/r_2 \tag{20}$$

and therefore

$$q_1 = (r_2/r_1)\, q_2 \tag{21}$$

where r_1/r_2 is the ratio of the outer to the inner tube radius. By introduction of the heat exchanging area on the primary side

$$A = S_1 \cdot L_T \tag{22}$$

where L_T is the tube length, Eqns. (18a) and (18b) may be written

$$dH_1/dA = -q_1 \quad ; \quad dH_2/dA_2 = q_2 \tag{23a, b}$$

where $dA = S_1\, dz$ is an increment of the heat exchanging area on the primary side and $dA_2 = S_2\, dz$ the corresponding increment on the secondary side. Since for tubes $A_2/A = r_2/r_1$ it follows

$$dH_2/dA = (r_2/r_1)\, q_2 = q_1 \tag{24}$$

In the following heat transfer coefficients α_1 and α_2 will be used which are related to the difference between bulk and wall temperature in the primary and secondary channel. They are defined by

$$q_1 = \alpha_1 (T_1 - T_{w1}) \quad ; \quad q_2 = \alpha_2 (T_{w2} - T_2) \tag{25a, b}$$

By using the solution of the steady state heat diffusion equation inside the tube wall, the wall temperatures may be eliminated resulting in the overall heat transfer relation

$$q_1 = \alpha_{eff} (T_1 - T_2) \tag{26}$$

with

$$1/\alpha_{eff} = 1/\alpha_1 + d_w^*/\lambda_w + 1/\alpha_2 \tag{27}$$

d_w^* is an effective thickness of the wall. It is equal to the wall thickness in case of a plane wall. For tubes it is given by

$$d_w^* = r_1 \ln (r_1/r_2). \tag{28}$$

3.3.1 Heat transfer without condensation

The energy balances for the primary and secondary circuit in a region where no condensation exists are given by

$$(mc_p)_1 (dT_1/dA) = - \alpha_{eff} (T_1 - T_2) \tag{29}$$

$$m_2 c_{p2} (dT_2/dA) = \alpha_{eff} (T_1 - T_2) , \tag{30}$$

with α_{eff} given by Eqn. (27).

Eqns. (29) and (30) can be combined to

$$d (T_1 - T_2)/dA = - \omega (T_1 - T_2) , \tag{31}$$

with

$$\omega = \alpha_{eff} [1/(mc_p)_1 + 1/(m_2 c_{p2})] . \tag{32}$$

For an increment ΔA of the primary heat exchanging area which is limited in such a manner that ω is practically constant within its boundaries A_i and A_{i-1}, Eqn. (31) can be integrated over this increment, giving

$$(T_1 - T_2)_i = (T_1 - T_2)_{i-1} e^{-\omega \Delta A} . \tag{33}$$

The energy balance Eqn. (15) can be used to eliminate $T_{1,i}$ from Eqn. (33) giving

$$T_{2,i} = T_{2,i-1} + \frac{\alpha_{eff}(1 - e^{-\omega\Delta A})}{m_2 \, c_2 \, \omega} (T_{1,i-1} - T_{2,i-1}) \cdot \qquad (34)$$

The corresponding primary temperature $T_{1,i}$ can then be obtained by substituting $T_{2,i}$ into the balance equation (15).

In that region of the primary channel where condensation is not possible, a correlation for convective heat transfer is used which is applicable to tubes, subject to cross flow of gas in the turbulent and laminar range. The correlation is given in appendix I. Materials properties of the gas-mixture entering the correlation are calculated from rules which are also quoted in this appendix.

The Nusselt number for a bundle of tubes is related to the Nusselt number Nu_1 for a single tube in the form

$$Nu_{bundle} = f_A \cdot Nu_1 \qquad (35)$$

where f_A is an adaptation factor which depends on the geometric arrangement of the tubes and is independent on fluid velocity and materials properties.

The factor f_A may be calculated from empirical correlations /8/. Typical values are in the range from about 1 to about 2.6. In this program module, however, f_A may also be determined for the initial state by precribing the gas outlet temperature in addition to the inlet temperature. This temperature may, for instance, be determined from a more sophisticated steam generator program. The additional temperature specification is compatible with the steam-generator model only for a certain value of its transfer function. For the initial state, therefore, f_A will be determined in such a manner that compatibility of the heat transfer function with the specified primary outlet temperature is attained. During a subsequent transient this value of f_A may be kept constant. The value of

f_A obtained in this way will normally deviate from that value which would have been obtained from correlations because this procedure also compensates at least partially simplifications in the program such as, for instance, the space independency of heat transfer coefficients. It may be regarded as a indication that the parameters of the simplified model are properly choosen if the value of f_A determined in this way does not deviate more that about 30% from values obtained from correlations.

3.3.2 Primary heat transfer with condensation

The heat transfer to the wall in the condensation region is determined on one hand by two heat resistances in series: that of the condensate film on the wall and that at the interface between bulk and film. On the other hand it is influenced by the deposition of latent heat in the film due to condensation (which acts as an additional heat transfer mechanism) and by heat exchange between the gas and "cold water droplets" which will be entrained from the film to the bulk. The latter effect is not modelled but may be taken into account by a correction factor in the expression for the energy transport to the condensate film.

The heat flux density on the primary side may be written in the form

$$q_1 = m_r \cdot L + \alpha_g (T_1 - T_f) , \qquad (36)$$

where m_r is a diffusion mass flow density from the bulk to the film. α_g is the heat transfer coefficient for the heat flow from the bulk to the film surface (with temperature T_f). m_r may be expressed by a mass transfer coefficient β_{wc} in the form

$$m_r = \beta_{wc} (p_s - p_{s,f}) , \qquad (37)$$

where $p_{s,f}$ is the steam partial pressure at the film surface.

Usual correlations for β_{wc} are given in appendix I. From the analogy between mass and energy transport follows, that β_{wc} and α_g are proportional. Defining the factor $\varkappa$ by

$$\beta_{wc} = \varkappa \alpha_g \quad , \tag{38}$$

the following expression for the heat flux density is obtained:

$$q_1 = \alpha_g [\varkappa \cdot L (p_s - p_{s,f}) + (T_1 - T_f)] \quad . \tag{39}$$

The difference of steam partial pressures appearing in this equation may by replaced by a corresponding temperature difference in the form

$$p_s - p_{s,f} = (dp_s/dT)_{sat} (T_1 - T_f) \quad , \tag{40}$$

if the bulk gas temperature is in equilibirium with the steam partial pressure. This applies to the condensation region with bulk condensation but not to the region with wall condensation only.

From (40) follows

$$q_1 = \alpha_g [1 + \varkappa \cdot L (dp_s/dT)_{sat}] (T_1 - T_f) \quad , \tag{41}$$

or

$$q_1 = \alpha_{f2} (T_1 - T_f) \quad . \tag{42}$$

With

$$\alpha_{f2} = \alpha_g [1 + \varkappa \cdot L (dp_s/dT)_{sat}] \tag{43}$$

This expression is replaced in the program by

$$\alpha_{f2} = \alpha_g [1 + a_c \varkappa L (\Delta p_s/\Delta T_{sat}(p_s))] \tag{44}$$

i.e. with the differential quotient replaced by a difference quotient and with a correction factor a_c included. This correction factor which should be equal or smaller than one, shall provide the possibility to take into account a possible energy transfer from the film to the bulk by entrainment of water droplets.

The difference quotient appearing in Eqn. (44) is evaluated from

$$\frac{p_s(T_1) - p_s(T_{w1})}{T_1 - T_{w1}} \tag{45}$$

using a function subroutine for the steam saturation pressure as a fuction of temperature.

The overall heat transfer coefficient α_1, defined by

$$q_1 = \alpha_1 (T_1 - T_{w1}) \tag{46}$$

is then given by

$$1/\alpha_1 = 1/\alpha_f + 1/\alpha_{f2} \quad , \tag{47}$$

where

$$\alpha_f = q_1/(T_f - T_{w1}) = \lambda_w/\delta_f \tag{48}$$

is the heat transfer coefficient of the condensate film.

In the condensation region I a different situation exists because the bulk gas temperature is not in equilibrium with the steam partial pressure. The local heat transfer is given by Eqn. (39) at the boundary only, which this region has in common with the condensation region II. On the other boundary (to the region without condensation), the latent heat contribution vanishes. In the condensation region I, therefore, the heat transfer coefficient α_{f2}, defined by Eqn. (42), varies from the value given by Eqn. (44) on one boundary to α_g at the other boundary. The integration of the energy equation over the condensation region I is performed with the arithmetic mean of the two boundary values taken as an average value for the heat transfer coefficient in this region.

Film heat transfer coefficients are evaluated in this module from empirical correlations proposed by Shekriladze and Gomelauri / 5 / which, by comparison with experimental data, have been checked to be applicable to banks of tubes with downward and horizontal flow. It is supposed, that these correlations are valid in all cases where the conditions for co-current film-gas flow are satisfied. These correlations are given in appendix I.[1)]

The correlation seems to predict, in general, lower values for the film thickness than the analysis given in ref. / 2 /, but these differences do not effect the result significantly. This is due to the fact that, under the flow conditions in question, the film heat transfer resistance forms only a small fraction of the total resistance between bulk and wall.

For the heat transfer coefficient α_g the same correlation is used which has been applied to the region without condensation. Since this correlation is based on the momentum-energy exchange analogy, this implies that the difference between wall shear stress and interfacial shear stress at the film surface is neglegted.

In the condensation region II, it follows from Eqns. (23a) and (16)

$$(mc_p)_1 \, (dT_1/dA) + L \;\; (dm_s/dA) \;\; = \;\; q_1 \;\; , \qquad (49)$$

with $(mc_p)_1$ given by Eqn. (14).

The second term on the left side of this equation represents the release of latent heat to the bulk flow. Bearing in mind that the change of the steam mass flow is assumed to be in accordance with thermal equilibrium, i.e. $T_1 = T_{sat}(p_s)$, the program module uses the balance equation (49) transformed as follows:
The derivative of m_s appearing in Eqn. (49) may be written

$$dm_s/dA = (dm_s/dp_s) \cdot (dp_s/dT)_{sat} \cdot (dT_1/dA) \qquad (50)$$

---

1) See also the comprehensive review of film correlations given in Ref. /4 /.

where the second factor on the right side of this equation is a derivative along the saturation line. From Eqn. (9) one obtains

$$dm_s/dp_s = m_{nc}(M_s/M_{nc})\ p_g/(p_g - p_s)^2 \qquad (51)$$

which, when inserted in Eqn. (50) gives

$$(mc_p)^*\ (dT_1/dA) = q_1 \qquad (52)$$

with

$$(mc_p)^* = (mc_p)_1 + m_{nc}(M_s/M_{nc})\ (p_g/(p_g - p_s)^2)\ (dp_s/dT)_{sat}\ L \qquad (53)$$

Eqn. (52) is a formal reduction of (49) to the form derived for the enthalpy flow without condensation. It must be kept in mind, however, that T_1 in this equation is supposed to be equal to the saturation temperature corresponding to the local steam partial pressure.

In the numerical treatment of the bulk condensation region an average value of $(mc_p)^*$ is assigned to this region which is chosen in the following manner:
The derivative $(dp_s/dT)_{sat}$ is replaced by the difference quotient

$$\frac{p_{sout} - p_{sin}}{T_{sat}(p_{sout}) - T_{sat}(p_{sin})} \qquad (54)$$

where p_{sout} and p_{sin} are the outlet and inlet steam partial pressures respectively. For p_s appearing in Eqn. (53) a weighted average of the inlet and outlet values is used

$$p_s = \gamma \cdot p_{sin} + (1 - \gamma)\ p_{sout} \qquad (55)$$

The weight factor γ has been determined by comparison with results of the detailed program described in ref. /2/ for the same boundary conditions. Optimal agreement was obtained with $\gamma = 0{,}6$.

3.3.3 Secondary heat transfer

In the subcooled region and the steam superheating region heat transfer coefficients evaluated from correlations for single phase convective heat transfer are used. In the nucleate boiling region a correlation of the type

$$T_{w2} - T_{sat} = c_{BO}\, q_2^{0.3} \tag{56}$$

is used where c_{BO} is a pressure dependent factor (see appendix I).

Since the fluid temperature in the evaporation region is equal to T_{sat} an effective heat transfer coefficient may be derived from

$$\alpha_{2BO} = q_2/(T_{w2} - T_{sat}) , \tag{57}$$

resulting in

$$\alpha_{2BO} = q_2^{0.7}/c_{BO} . \tag{58}$$

In order to obtain a smooth transition of convection and boiling heat transfer the following formula is used for the effective heat transfer coefficient in the nucleate boiling region

$$\alpha_2 = [\alpha_{2BO}^2 + \alpha_{2conv}^2]^{1/2} , \tag{59}$$

where α_{2conv} is the heat transfer coefficient in the water region.

Empirical correlations for the heat transfer in the film boiling region and for the DNB-quality are given in appendix I.

Since the heat capacity of water is not independent on temperature, effective heat capacities are calculated from difference quotients as follows:

For the water region c_p is calculated from

$$c_{pw} = (h_{2in} - h_{satw})/(T_{2in} - T_{sat}) \, , \tag{60}$$

if the inlet enthalpy h_{2in} is smaller than h_{satw}. The inlet temperature and the saturation temperature of water, therefore, are used as supporting points for interpolations when relating enthalpies to temperatures in this region.

For the steam region the effective c_p is calculated from the difference quotient

$$c_{ps} = (h_{sc} - h_{sats})/(T_{sc} - T_{sat}) \tag{61}$$

where h_{sc} is a suitable enthalpy in the steam region ($h_{sc} > h_{sats}$) and T_{sc} the corresponding temperature. The secondary outlet enthalpy is taken for h_{sc} if this enthalpy is known in advance and belonging to the steam region.

4. Solution of the steam generator equations

4.1 Determination of interval boundaries

The interval boundaries determined by the flow condition on the secondary side may be obtained by solving Eqn. (34) explicitely for ΔA as follows:

$$\Delta A = -\frac{1}{\omega} \ln \left[1 - \frac{(h_{2,i} - h_{2,i-1})\, m_2\, \omega}{\alpha_{eff}\, (T_{1,i-1} - T_{2,i-1})}\right] \tag{62}$$

where the secondary temperature difference has been replaced by the corresponding enthalpy difference.

If $h_{2,i}$ is a specified enthalpy then ΔA from Eqn. (62) is the area increment necessary to advance the enthalpy from $h_{2,i-1}$ to $h_{2,i}$. If $h_{2,i}$ and $h_{2,i-1}$ are enthalpies either representing region boundaries or the inlet or outlet plane, then this equation can be used to determine the extent of the subcooled, the evaporation and the steam superheating region.

The primary gas temperature at these region boundaries can be calculated from the energy balance Eqn. (15) if the boundary is outside of the region of primary bulk condensation.

The same formalism can be used to determine the locations of the boundaries of the condensation regions on the primary side if the energy balances are used which have been transformed to the same type of equations which govern the regions without condensation.

If a bulk condensation region exists then its boundary on one side will be the plane of the secondary inlet and the boundary on the other side that to the condensation region I (see Fig. 1). The area occupied by this region may be evaluated from Eqn. (62) with

$$h_{2,i-1} = h_{2in} \quad \text{and} \quad h_{2,i} = h_{2B} ,$$

where h_{2in} is the secondary inlet enthalpy and h_{2B} the secondary enthalpy at the other boundary. ω has to be evaluated using Eqn. (32) with $(mc_p)_1$ replaced by $(mc_p)^*$ according to Eqn. (53) and α_{eff} given by Eqn. (47).

The enthalpy h_{2B} has to be calculated from the energy balance (15) with $T_{1,i}$ equal to the saturation temperature corresponding to the steam partial pressure p_{s1} at the location of the boundary:

$$h_{2B} = h_{2in} - (mc_p)^* \, (T_{sat}(p_{s1}) - T_{1out})/m_2 . \qquad (63)$$

Since the loss of steam in the bulk due to film formation is neglected in condensation region I, p_{s1} may be set equal to the inlet steam partial pressure. We note that the assumption of thermal equilibrium in this region implies that

$$T_{1out} = T_{sat}(p_{sout}) . \qquad (64)$$

If this condensation region exists, it follows that the outlet steam partial pressure p_{sout} must be smaller then the inlet value p_{sin}.

If the condensation region I exists, two cases have to be considered: If the condensation region II also exists, then the boundary which is common to both regions lies inside the primary channel. If this is not true, the left boundary of this condensation region is identical with the inlet plane of the secondary flow.

If Eqn. (62) is used to determine the extent of this region, then $h_{2,i-1}$ has to be replaced by the boundary enthalpy h_{2B} in the first case and by h_{2in} in the second case. The other boundary is determined by that point, where the primary wall temperature becomes equal to $T_{sat}(p_{sin})$.

In the following enthalpies and temperatures at the left boundary will be characterized by the index n and those at the right boundary (common to the region without condensation) by n+1. Thus

$$\begin{aligned} h_{2,i-1} &= h_{2,n} = h_{2B} \text{ in case 1, or } = h_{2in} \text{ in case 2 ,} \\ T_{1,i-1} &= T_{1,n} = T_{1B} \text{ in case 1, or } = T_{1in} \text{ in case 2 ,} \end{aligned} \tag{65}$$

where T_{1B} is the primary temperature at the boundary.
From the energy balance Eqn. (15) follows

$$h_{2,n+1} = h_{2,n} - (mc_p)_1 \, (T_{1,n+1} - T_{1,n})/m_2 \, . \tag{66}$$

The corresponding gas temperature at the boundary is given by

$$T_{1,n+1} = T_{sat}(p_{sin}) + q_{1,n+1}/\alpha_{1,n+1} \tag{67}$$

if $q_{1,n+1}$ is the local heat flux density and $\alpha_{1,n+1}$ the primary heat transfer coefficient at the boundary. Since at this point no latent heat is released, the single phase heat transfer coefficient valid for the adjacent region without condensation has to be taken.

To eliminate the heat flux density $q_{1,n+1}$, the equation for the energy transfer to the secondary side

$$q_{1,n+1} = \alpha_{eff,n+1} (T_{1,n+1} - T_{2,n+1}) \tag{68}$$

may be used, where $\alpha_{eff,n+1}$ is the local effective heat transfer coefficient according to Eqn. (27). The gas temperature at the boundary may then be written:

$$T_{1,n+1} = (\alpha_{1,n+1} T_{sat}(p_{sin}) + \alpha_{eff,n+1} T_{2,n+1})/(\alpha_{1,n+1} - \alpha_{eff,n+1}). \tag{69}$$

By inserting (69) into Eqn. (66), the following expression for the secondary enthalpy difference corresponding to the extent of the region is obtained:

$$h_{2,n+1} - h_{2,n} = \frac{(mc_p)_1 (\alpha_{1,n+1} (T_{sat}(p_{sin}) - T_{1,n}) - \alpha_{eff,n+1} (T_{1,n} - T_{2,n}))}{(mc_p)_1 \alpha_{eff,n+1}/c_{pw} + (\alpha_{eff,n+1} - \alpha_{1,n+1}) \cdot m_2} \tag{70}$$

The extent of the heat exchanging area which is occupied by this condensation region is obtained, if this expression is inserted into Eqn. (62).

4.2 Boundary conditions

The solution of the steam generator equations with countercurrent flow can be achieved with some suitable iteration procedure only. The method explained below is illustrated by the flow chart given as Fig 3.

The solution routine starts at the inlet plane of secondary flow and proceeds in the direction to the secondary outlet. This means that the integration on the primary side starts at the primary outlet and proceeds opposite to the primary flow direction. Numerical experiments have shown, that a reverse integration method starting at the secondary outlet plane shows a less favourable behaviour and requires, in general, a larger computational effort.

Since the primary outlet temperature and the corresponding steam partial pressure are not known in advance because they are dependent quantities to be determined, the solution procedure starts with some initial guess.

The program determines the type of the first region with regard to the secondary flow from the specified secondary inlet enthalpy and with regard to the primary flow from the assumed primary outlet temperature. If the outlet temperature is lower then $T_{sat}(p_{sin})$, then the first region is a condensation region of type II and the outlet steam partial pressure is equal to the corresponding saturation pressure. If the outlet temperature is larger or equal to this saturation temperature, but the primary wall temperature lower, then the first region is a condensation region of type I. If both temperatures are above the saturation temperature, no condensation on the primary side exists.

The routine then evaluates the area increment to attain the next region boundary. If this increment is larger than the available heat exchanging area this region is identified as the final region and the program will calculate the outlet conditions using Eqns. (34) and (15). If the available area is larger then the area required by the first region then the program will calculate the temperatures and the heat fluxes at the upper region boundary. It evaluates the area available for the next regions by subtracting the area consumed by the region just treated from the previous available area and proceeds to calculate the next region boundary. In this way a "do-loop" is formed, which is terminated if either the area increment is larger then the available area or if the last area is the steam region on the secondary side (which has an unlimited area increment).

If the first guess for the primary outlet temperature is not correct by chance then the evaluated primary temperature at the end of the final region will deviate from the prescribed gas inlet temperature. The program initiates in this case an iteration procedure which modifies the gas outlet temperature and repeats the evaluation procedure until the prescribed boundary conditions are satisfied. Convergence is attained and the iteration terminated if the absolute value of the difference

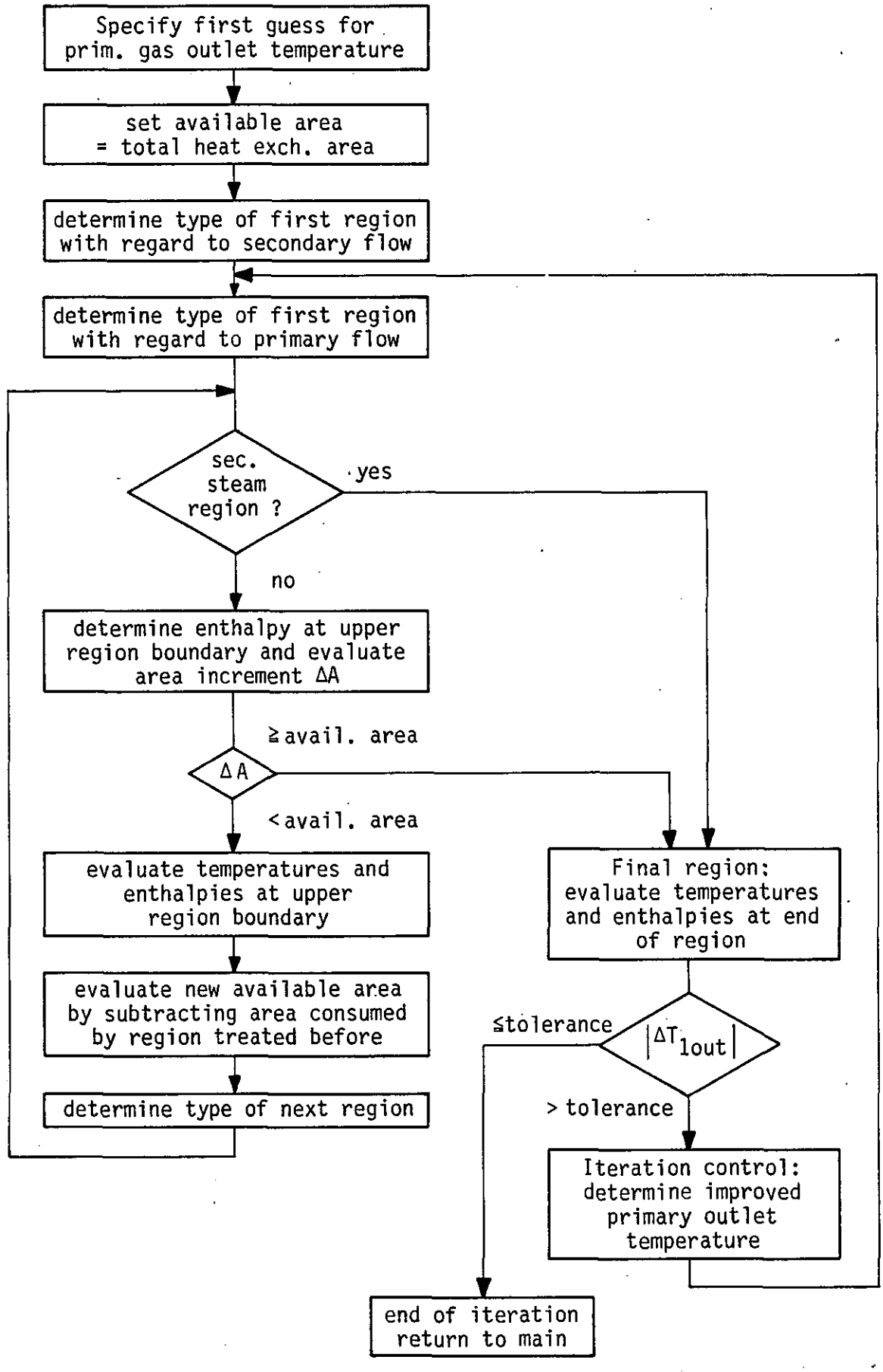

Fig. 3: Flow chart of variable boundary evaluation scheme

between the prescribed gas inlet temperature and the evaluated primary temperature at the end of the final section becomes less then a specified error tolerance.

The iteration process is controlled by a general purpose iteration subroutine. This routine determines iteratively a root of a steady function within a prescribed interval of its argument. It is based on a search step procedure which changes over to linear interpolation when the root has been bracketed. The same iteration procedure is used for determining the heat transfer adaption factor f_A (Eqn. (35)). In this case f_A will be modified until the evaluated and specified gas temperatures fit together.

5. Sample calculations

To demonstrate the qualitative effect of primary condensation, comparative calculations have been performed for identical steam generator geometry and boundary conditions but once with the simulation of primary condensation included and once with the corresponding part of the program bypassed.

Figs. 4, 5 and 6 show the results of these calculations. Full lines refer to condensation included and broken lines to condensation neglected. The primary gas is assumed to be a mixture of helium and steam. The geometry data and boundary condition underlying the results are listed in table 1.

Fig. 4 shows the temperature distribution in a steam generator with a steam fraction of 50% at the inlet of the primary channel. These distributions have been generated with the program version described in ref. /2/ with a subdivision of the heat exchanging area into 50 discretisation intervals. The figure shows that, although the primary condensation exists in a small fraction of the heat exchanging area only, a strong effect on the overall temperature distributions is observed.

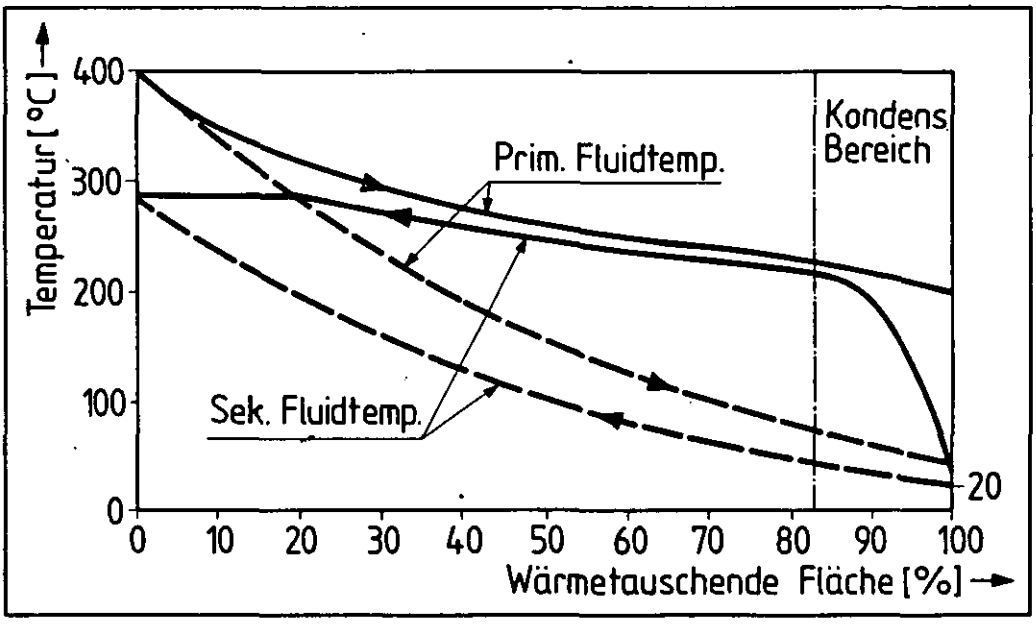

Fig. 4: Influence of primary condensation on temperature distributions in a steam generator

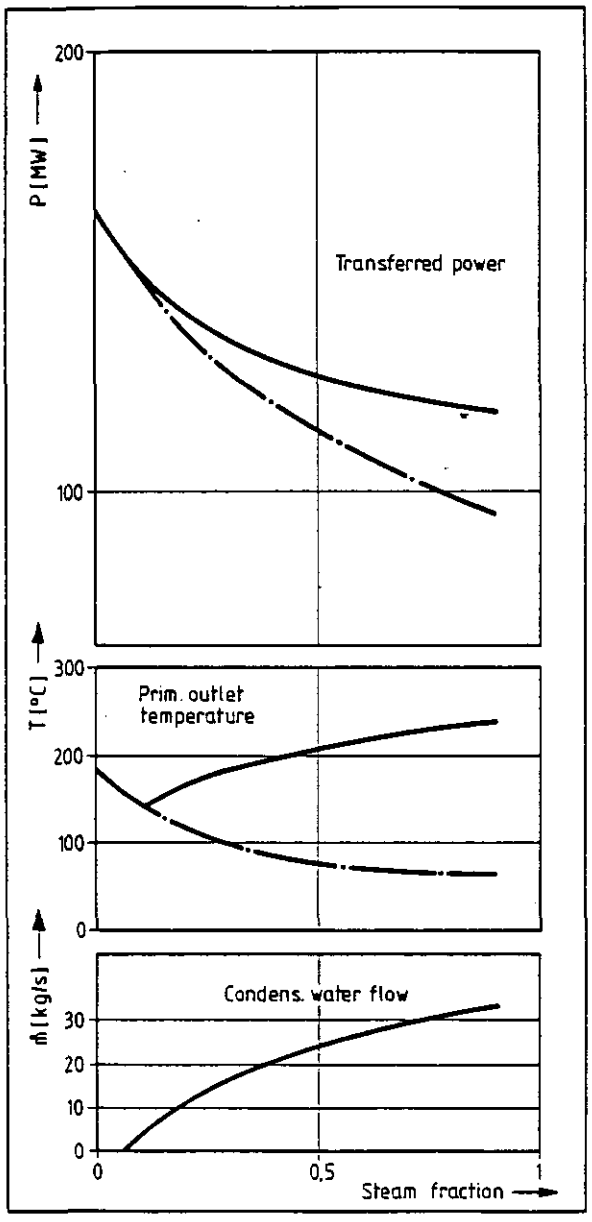

Fig. 4: Influence of primary steam fraction for constant primary inlet mass flow

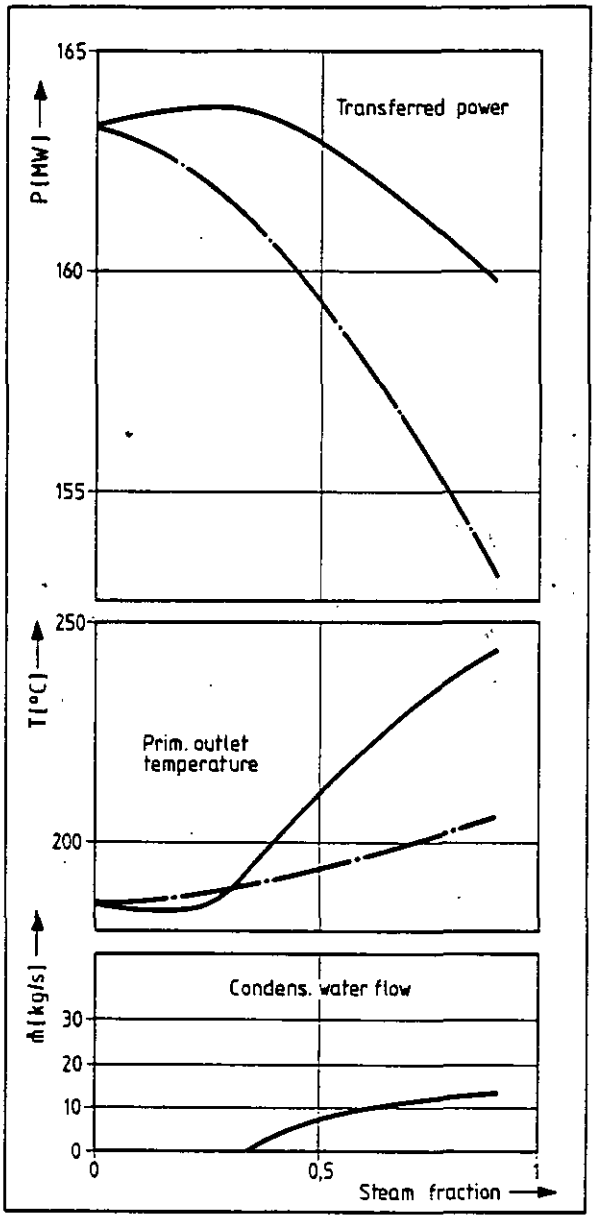

Fig. 5: Influence of primary steam fraction for constant primary inlet enthalpy flow

Fig. 5 shows the power transfer, the primary outlet temperature and the-mass flow of condensate water if the steam fraction at the primary inlet is varied but the primary mass flow kept constant. The primary inlet temperature and all other parameters (listed in table 1) are also kept constant.

In this case the power transfer decreases continuously with inceasing steam fraction. The reason is that the effective specific heat capacity of the mixture decreases with increasing steam content because the specific heat capacity of steam is about two times lower than that of helium. Thus, with constant inlet mass flow and temperature the enthalpy inflow decreases with increasing steam fraction. The primary condensation increases the transmitted power. The primary outlet temperature first decreases with low steam inflow but with the incipience of bulk condensation it attains the saturation temperature of steam which increases with increasing steam partial pressure at the outlet.

Fig. 6 shows the situation if the enthalpy flow at the primary inlet is kept constant by proper adjustment of the mass flow. The primary inlet temperature is again kept constant. The continuous decrease of the power transfer without condensation is obviously mainly due to the dependence of the primary heat transfer coefficient on changing effective gas properties and mass flow. Enhanced heat transfer due to film condensation compensates this effect with moderate steam fractions causing a slight increase of the power transfer. With the onset of bulk condensation, however, the transmitted power passes though a maximum and decreases with increasing steam fraction.

Table 1

Geometry data and boundary conditions for sample calculations

Fig.	4	5	6
Primary heat exchanging area (m^2)	925	1000	1000
Tube outer diameter (m)	0,025	0,025	0,025
Tube inner diameter (m)	0,019	0,019	0,019
Number of parallel tubes	80	80	80
Primary side			
total free flow area (m^2)	1,65	1,65	1,65
inlet temperature (°C)	400	500	500
mass flow (kg/s)	30	100	var.
total pressure (bar)	40	40	40
inlet steam pressure (bar)	20	var.	var.
Secondary side			
inlet temperature (°C)	20	20	20
pressure (bar)	80	215	215
mass flow (kg/s)	20	60	60

Appendix I

Correlations

1. Heat transfer coefficient for cross flow to tubes

The following correlation covering the laminar and turbulent flow range is used (V. Gnielinski, ref. /3/, Ge1/Ge3)

$$Nu_1 = 0{,}3 + [Nu_{lam}^2 + Nu_{turb}^2]^{1/2} \quad , \qquad (A\ 1)$$

$$\alpha_1 = \frac{\lambda_g}{d_H} Nu_1 \qquad (A\ 2)$$

with

$$Nu_{lam} = 0{,}664 \cdot Re^{1/2}\ Pr^{1/3} \quad , \qquad (A\ 3)$$

$$Nu_{turb} = c_f\ Re\ Pr/(1 + \sqrt{c_f}\ 12{,}7\ (Pr^{2/3} - 1)) \qquad (A\ 4)$$

and

$$c_f = 0.037\ Re^{-0{,}2} \qquad (A\ 5)$$

The Reynolds number has to be formed with a characteristic length $d_H = \frac{\pi}{2} d_1$ where d_1 is the outer tube diameter.

2. Single phase heat transfer to the secondary flow

In the turbulent range the following correlation for flow inside tubes is used (V. Gnielinski, ref. /3/, Gb3)

$$Nu = c_f\ (Re - 1000)\ Pr/(1 + \sqrt{c_f}\ 12{,}7\ (Pr^{2/3} - 1)) \qquad (A\ 6)$$

with c_f given by Eqn. (A 5). (A 6) is a modification of correlation (A 4) which extends the applicability of the correlation to the transition range to laminar flow. The inner tube diameter has to be used as characteristic length.

The program limits Nu at low flow velocities to a minimum value of 3,66 which is the asymptotic value of the Nusselt number for laminar flow in tubes. The heat transfer coefficients generated by the program from these correlations are multiplied by correction factors which may be specified by input. Thus the possibility is provided to adjust the heat transfer coefficient, for instance, to experimental data. We note that results for a gas-water steam generator are normally not very sensitive to these heat transfer coefficients because the overall heat transfer resistance between the primary and the secondary flow is mainly determined by the primary heat transfer coefficient.

3. Nucleate boiling heat transfer

The nucleate boiling correlation Eqn. (56) is one of the numerous correlations existing. It is proposed by F. Müller /6/ and fits experimental data in the following range of parameters:

pressure	1,52	to	181	bar
flow velocity	0,65	to	6,0	m/s

For the pressure dependent factor c_{BO} the following empirical relation has been proposed

$$c_{BO} = 1/(c_o \cdot p_r^{0,23} (1 + 4,66\, p_r^{5,33})) \qquad \text{(A 7)}$$

with $c_o = 7,5\ (W/m^2)^{0,3}\ K^{-1}$ and $p_r = p/p_{crit} \cdot p_{crit}$ is the critical pressure of water (221,2 bar).

4. Film boiling heat transfer

It is assumed that in the film boiling region a liquid deficient flow regime exists. The following correlation is used

$$\alpha_{DNB} = \frac{\lambda_s}{d_1} Nu_{DNB} \quad , \qquad (A\ 8)$$

$$Nu_{DNB} = 1.09 \cdot 10^{-3} Re_s^{*\,0,989} Pr_{s,w}^{1,41} Y^{-1,15} \qquad (A\ 9)$$

with

$$Y = 1 - 0,01 \left[\left(\frac{\rho_w}{\rho_s} - 1 \right) (1 - x) \right]^{0.4} \qquad (A\ 10)$$

and

$$Re_s^* = Re_s \left[x + (\rho_s/\rho_w) (1 - x) \right] . \qquad (A\ 11)$$

The Reynolds number Re_s has to be evaluated with steam properties at bulk fluid temperature and the Prandtl number with steam properties at wall temperature. The type of correlation has been proposed by S. Miropolskii. The specific form given above has been developed by D.C. Groeneveld /6/ using experimental results in the following parameter range:

pressure	68	to	215	bar
mass flow density	700	to	5300	kg/m² s
steam quality	0,10	to	0,90	
heat flux density	120	to	2100	kW/m²

The correlation is not applicable outside this range. The program evaluates α_{DNB} at the average quality $\bar{x} = 0,5\ (1 + x_{DNB})$.

5. Heat transfer across the condensate film

The following correlation applicable to condensation outside horizontal tubes is used

$$\alpha_F = [\frac{1}{2}\alpha_S^2 + [\frac{1}{4}\alpha_S^4 + \alpha_G^4]^{1/2}]^{1/2} \quad , \qquad \text{(A 12)}$$

where α_S is a contribution due to shear stress and α_G due to gravitational forces.

$$\alpha_G = 0{,}725\ [\frac{\rho_w(\rho_w - \rho_s)\ g \cdot L\ \lambda_w^3}{d_1\ \mu_w\ (T_1 - T_f)}]^{1/4} \qquad \text{(A 13)}$$

$$\alpha_S = \frac{\lambda_w}{d_1} Nu_F \quad ; \quad Nu_F = 0{,}9 \cdot Re_F^{0,5} \qquad \text{(A 14)}$$

and

$$Re_F = \frac{\rho_w\ v_g\ d_1}{\mu_w} \quad . \qquad \text{(A 15)}$$

v_g is the average gas velocity between tubes and g the gravitational acceleration constant.

The factor 0,9 in (A 14) is valid in that range where are no boundary layer separation occurs. At high velocities with boundary layer separation this factor has to be replaced by 0,59. The correlation has been derived by Shekriladze and Gomelauri /5/ for downward flow. The correlation is in agreement with experimental data for downward and horizontal flow. It is supposed that it is also applicable to upward flow if $\alpha_S \gg \alpha_G$.

6. Quality at departure from nucleate boiling

Since the total power transfer is not very sensitive to the extent of the evaporation region with DNB conditions a simplified treatment of the phenomenon is justified. The correlation used for x_{DNB} is actually a correlation for dry-out at the heat exchanging wall. Heat flux densities exceeding the critical value are not expected to occur.

x_{DNB} is calculated from

$$x_{DNB} = \Phi(p) \left(\frac{w_o}{w}\right)^{0,5} \left(\frac{d_o}{d_2}\right)^{0,15} \qquad \text{(A 16)}$$

where w is the mass flow density and d_2 the inner tube diameter. Φ is a pressure dependent function given by

$$\Phi = 0,39 + 3,53 \cdot p_r - 10,3 \cdot p_r^2 + 7,62 \cdot p_r^3 \qquad \text{(A 17)}$$

with $p_r = p/p_{crit}$. The reference mass flow w_o is equal to 1000 kg/m²s and the reference diameter d_o equal to $8 \cdot 10^{-3}$ m.

The correlation is empirical and has been developed by Doroschuk, Levitan and Lantzman /7/. It is applicable to the following range of parameters:

mass flow density	750	to	3000 kg/m² s
pressure	10	to	167 bar

The correlation can probably be extended to higher pressures with the function Φ kept constant above 150 bar.

7. Mass transfer with condensation

From the analogy between mass and energy transfer follows for small mass transfer rates

$$\beta_{wc} = \varkappa \alpha_g \qquad \text{(A 18)}$$

with

$$\varkappa = \frac{R \cdot T_1 M_s}{c_{pg} p_g \overline{M}} \left(\frac{Pr}{Sc}\right)^{2/3} \qquad \text{(A 19)}$$

$\overline{M}$ is the effective molecular weight and c_{pg} the specific heat capacity of the gas mixture. Sc is the Schmidt number for diffusion of steam in the noncondensable gas:

$$Sc = \nu/D_m \qquad \text{(A 20)}$$

where ν is the kinematic viscosity and D_m the mass diffusion coefficient. The latter value is evaluated from the following expression

$$D_m = 0{,}64 \cdot 10^{-8} T_1^{1{,}68}/p_g \qquad \text{(A 21)}$$

which is predicted by the Chapman-Enskog theory for the diffusion of steam in helium. Eqn. (A 21) gives the diffusion coefficient in m^2/s if p_g is given in bar and T_1 in Kelvin.

If β_{wc} is used in connection with Eqn. (37) an improvement can be achieved if p_g in Eqn. (A 19) is replaced by

$$p_{log} = (p_s - p_{s,f})/\ln((p_g - p_{s,f})/(p_g - p_s)) \qquad \text{(A 22)}$$

8. Rules for evaluating mixture thermal properties of the primary gas mixture

The specific heat capacity $\bar{c}_p$ of the gas mixture is calculated as a mass flow weighted quantity

$$\bar{c}_p = \frac{c_{pnc}\, m_{nc} + c_{ps}\, m_s}{m_{nc} + m_s} \qquad \text{(A 23)}$$

The dynamic viscosity of the gas mixture is calculated from

$$\bar{\mu} = \frac{\sqrt{M_{nc}}\; p_{nc}\, \mu_{nc} + \sqrt{M_s}\; p_s\, \mu_s}{\sqrt{M_{nc}}\; p_{nc} + \sqrt{M_s}\; p_s} \qquad \text{(A 24)}$$

The thermal conductivity of the gas mixture is calculated from

$$\bar{\lambda} = \varphi(y_{nc}\, \lambda_{nc} + y_s\, \lambda_s) + (1 - \varphi)\left(\frac{y_{nc}}{\lambda_{nc}} + \frac{y_s}{\lambda_s}\right) \qquad \text{(A 25)}$$

with

$$y_{nc} = p_{nc}/p_g \quad ; \quad y_s = p_s/p_g \qquad \text{(A 26)}$$

and

$$\varphi = 0{,}3 + 0{,}4 \cdot y_{nc} \qquad \text{(A 27)}$$

Appendix II

Instructions for use of the program

The steam generator module described in this report is written as a FORTRAN subroutine

SUBROUTINE CCORDE(N)

The data exchange beween the module and the calling program is achieved via a common block named DECOM. The module can be used to simulate several steam generators with different geometry and different operating conditions. All variables, therefore, which define the steam generator design and the boundary conditions are arrays. N is the identification number of the steam generator. The Nth variable of each array is assigned to the steam generator with identification number N.

Common block data:

The following quantities have to be supplied by the calling program:

WSEK	secondary mass flow (kg/s),
PSEK	secondary pressure (bar),
ENTIN	secondary inlet fluid enthalpy (J/kg),
TFIN	secondary fluid inlet temperature corresponding to ENTIN (K),
ENTSTM	steam enthalpy h_{sc} (Eqn. (61)) to be used for evaluation of c_{ps},
TSTM	steam temperature corresponding to ENTSTM (K),
WPRIM	primary mass flow (kg/s), (positive sign for cocurrent flow, negative sign for countercurrent flow),
PG	primary total pressure (bar),
TGIN	primary inlet temperature (K),
TGOUT	primary outlet temperature (prescribed value for KIT=2, first guess for KIT=1),

PDIN steam partial pressure at primary inlet (bar),

WWIN water mass flow at primary inlet (kg/s),

AREAH total heat exchanging area (m^2),

AREAG primary free flow area (m^2),

AREAS total secondary flow area (= inner tube flow area times number of parallel tubes),

DIAM outer tube diameter (m),

DIAI inner tube diameter,

GHTC factor f_A (Eqn. (35))
(prescribed value for KIT≠2, first guess for KIT=2),

ACORR correction factor a_c in Eqn. (44),

FHCW adaption multiplier for the heat transfer coefficient for water flow in the secondary channel, (see App. I, sect. 2),

FHCD dto for steam flow,

CDWA thermal conductivity of the heat exchanging wall (W/m K),

CPNC effective specific heat capacity at constant pressure for the mixture of noncondensable gases in the primary channel (W/kg K) at primary inlet temperature,

CPW1 dto for water at saturation temperature corresponding to the inlet steam partial pressure,

CPD1 dto for steam at primary inlet temperature,

CDNC effective thermal conductivity of the mixture of noncondensable gases in the primary channel (W/m K) at primary inlet temperature,

CDW1 dto for water at primary inlet saturation temperature,

CDS1 dto for steam at primary inlet temperature,

VISNC dynamic viscosity of the mixture of noncondensable gases (Kg/s m) at primary inlet temperature,

VISW1 dto for water at primary inlet saturation temperature,

VISS1 dto for steam at primary inlet temperature,

GMOL effective molecular weight of the mixture of noncondensable gases,

DMOL effective molecular weight of steam at primary pressure, and average primary gas temperature,

ROW water density in the secondary flow (kg/m^3) at secondary pressure and inlet temperature,

PRANDW Prandtl number of water in the secondary flow at the average of the inlet and saturation temperature,

PRANDS dto for steam at the average of outlet and saturation temperature,

CDW2 thermal conductivity of water for the secondary flow (W/m K) at the average of inlet and saturation temperature,

CDS2 dto for steam at the average of outlet and saturation temperature,

VISW2 dynamic viscosity of water in the secondary flow at same temperature as for CDW2,

VISS2 dto for steam at same temperature as for CDS2,

ERMAXT temperature error tolerance for iterations (recommended: 0.1 K),

ITMAX maximum number of iterations allowed,

ITPR print control index for iterations (the iteration process will be listed if ITPR is set $\neq$ 0),

KIT iteration control index

KIT=0 no iteration will be initiated,

KIT=1 the primary outlet temperature for counter current flow will iteratively be determined,

KIT=2 GHTC will iteratively be determined according to a prescribed outlet temperature.

NP number of region boundaries existing (= number of regions + 1).

Return variables generated by the program:

ENTOUT	secondary outlet enthalpy (J/kg),
TFOUT	secondary outlet temperature (K),
TGOUT	primary outlet temperature (generated for KIT≠2),
WDIN	primary inlet steam mass flow (kg/s),
WDOUT	primary outlet steam mass flow,
WWOUT	condensate water mass flow at primary outlet (kg/s),
DEPOW	transferred power (W),
GHTC	factor f_A resulting from an iteration with KIT=2,

Arrays for displaying variables at region boundaries (K=1 to NP(N)).

ARH(K)	heat exchanging area from inlet to boundary with index K,
DAR(K)	region area,
TG(K)	primary fluid temperature,
TF(K)	secondary fluid temperature,
TW1(K)	primary wall temperature,
TW2(K)	secondary wall temperature,
X(K)	steam quality
HFX(K)	primary heat flux density,
HCP(K)	primary heat transfer coefficient,
HCS(K)	secondary heat transfer coefficient,
ENTX(K)	secondary fluid enthalpy,
WGCP(K)	quantity $(mc_p)^*$ for the primary flow

CCORDE calls the following auxiliary subroutines (SR) and functions (F):

F	TPSAT(P)	saturation temperature of water as a function of pressure,
SR	ENSAT(ES,P)	saturation enthalpy of water as a function of pressure,
F	EVAP(P)	latent heat of water as a function of pressure,
SR	HTC1(CNUS,R,PR)	Nusselt number as a function of Reynolds and Prandtl number for cross flow to tubes,
SR	HTC2(CNUS,R,PR)	dto for flow inside tubes,
SR	DNB(XDNB,P,G,D)	DNB quality as a function of pressure, mass flow density and diameter,
SR	HTDNB(CNUS,R,PR,ROW,ROS,X)	Nusselt number for DNB heat transfer as a function of Reynolds and Prandtl number, water and steam density and quality,
SR	ITER(F,X,...)	routine for iterative determination of a root of a function F(x).

Nomenclature

A	primary heat exchanging area,
A_2	secondary heat exchanging area,
a_c	correction factor in Eqn. (44),
c_{pw1}	specific heat capacity for water in the primary flow,
c_{ps1}	specific heat capacity for steam in the primary flow,
c_{pnc1}	effective specific heat capacity for the mixture of noncondensable gases in the primary flow,
c_{pw}	effective specific heat capacity for water in the secondary flow,
c_{ps}	effective specific heat capacity for steam in the secondary flow,
c_{BO}	nucleate boiling coefficient (Eqn. (56)),
d_w^*	effective wall thickness of the heat exchanging wall,
D_m	mass diffusion constant,
f_A	heat transfer adaptation factor for tube bundles,
H	enthalpy flow,
h	fluid enthalpy,
h_{satw}	saturation enthalpy of water in the secondary flow,
h_{sats}	saturation enthalpy of steam in the secondary flow,
h_s	primary steam enthalpy,
h_{1w}	primary bulk water enthalpy,
h_{1wf}	primary film water enthalpy,
L	latent heat of evaporation,
m	total mass flow,
m_s	primary steam mass flow,
m_w	primary bulk water flow,
m_f	primary film water flow,
m_{H_2O}	total primary water flow,
m_{nc}	primary mass flow of noncondensable gases,

m_r primary diffusion mass flow,

$(mc_p)_1$ quantity defined by Eqn. (14),

$(mc_p)^*$ quantity defined by Eqn. (53),

M_s effective molecular weight of steam,

M_{nc} effective molecular weight of the mixture of noncondensable gases,

Nu Nusselt number,

p_g primary total pressure,

p_{nc} primary partial pressure of noncondensable gases,

p_s primary steam partial pressure in the bulk flow,

$p_{s,f}$ steam partial pressure at the film surface,

Pr Prandtl number,

q heat flux density,

R universal gas constant,

r_1, r_2 outer and inner radius of heat exchanging tubes respectively,

Re Reynolds number,

S heat exchanging parameter of flow channel,

T bulk fluid temperature,

T_f film surface temperature,

T_w wall temperature,

T_{sat} saturation temperature,

x steam quality,

α heat transfer coefficient with regard to bulk and wall temperature,

α_g heat transfer coefficient with regard to bulk and film temperature,

α_{eff} effective heat transfer coefficient with regard to the primary and secondary bulk fluid temperature,

β_{wc} mass transfer coefficient in the primary bulk flow,

δ_f film thickness,

λ_w thermal conductivity of the heat exchanging wall,

μ dynamic viscosity,

ν kinematic viscosity,

$\varkappa$ ratio of mass and heat transfer coefficient,

ω quantity defined by Eqn. (32),

ρ density.

Subscripts:

1 primary flow,

2 secondary flow,

g primary gas mixture,

w water,

s steam,

in inlet,

out outlet.

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